HIGH SPEED OIL-FREE TURBOMACHINERY STRING

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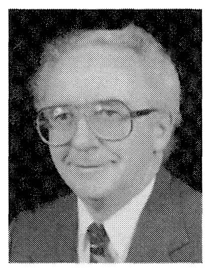

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ABSTRACT

This paper is a discussion of the design, manufacture, test, and field operation of a totally oil-free string of rotating equipment. The turbomachinery string consists of a multistage centrifugal compressor driven by a single valve multistage steam turbine. The steam turbine is rated at 1837 kW (2463 hp) with a maximum continuous speed of 9734 rpm. The string is currently operating in propane refrigeration service at a petrochemical complex in Saudi Arabia. Both the steam turbine and the compressor have magnetic journal and thrust bearings, dry gas face seals to contain the process gas in the compressor, and pneumatic/electronic actuators control the steam flow through the steam turbine while providing trip protection.

The design of the compressor, steam turbine, associated components, and the control theory required by the magnetic bearings are reviewed and discussed in detail. The development of the theoretical rotordynamic simulation for magnetic bearings is presented and the results are compared with actual operating conditions. Special consideration is given to the coastdown case where the rotors are delevitated onto the auxiliary bearings. Shop testing and control system tuning of the assembled turbomachinery string along with field operating experience are reviewed.

Finally, the trouble spots encountered during the design, manufacture, testing, installation, and field operation of this oil-free string are discussed and suggestions are made in order to avoid these problems in the future.

INTRODUCTION

Previous applications of magnetic bearings on rotating turbomachinery strings have proven successful: one such installation has magnetic bearings on the compressor and oil lubricated bearings on the electric motor and gear; another installation has magnetic bearings on the compressor and oil lubricated bearings on the steam turbine (DeChoudhury, et al., 1993). These applications, where magnetic bearings have been applied only to the compressor component of the turbomachinery string, still require the use of an oil lubrication system for the driver. The turbomachinery string to be discussed in this presentation has been designed to completely eliminate the oil system, creating a completely oil-free string of turbomachinery. The string consists of a single valve multistage steam turbine coupled to a multistage centrifugal compressor and is installed at the Saudi Petrochemical Company's petrochemical complex in Al Jubail, Saudi Arabia (Figure 1).

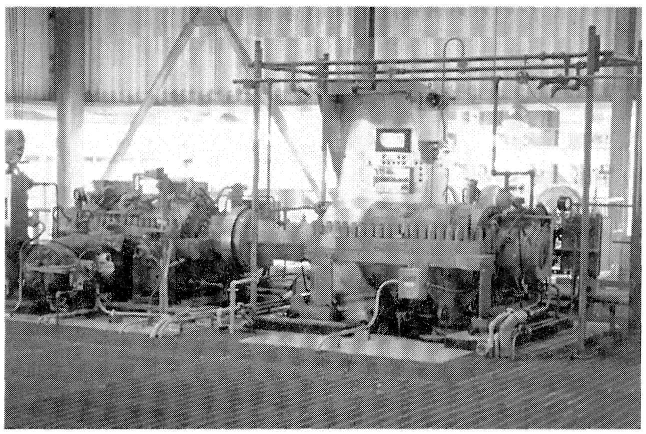

Figure 1. Oil-Free Turbocompressor String.

In the past, turbomachinery strings have required complex lubricating, seal, and control oil systems for effective operation. Compressors have typically used oil for their journal bearings, thrust bearings, and seals. Steam turbines have required oil, not only for their journal and thrust bearings, but also for the operation of the servomotor, trip and throttle valve, overspeed trip mechanism, and governor. Since these auxiliary systems require oil supplied at various pressures and flows, the resulting complex oil systems have made the option of an oil-free turbomachinery string very attractive. Several recent equipment and auxiliary system advances have made this possible:

- The use of dry gas face seals in compressors has achieved common acceptance over the last several years and has helped justify the elimination of the oil type seal and the seal oil system.
- On steam turbines, electronic governors operating solely on electronic signals generated by shaft speed detection probes have replaced hydraulic governors, due to their simplicity of design and overall reliability. Overspeed trip mechanisms are almost exclusively electronic devices utilizing electronic signals generated by multiple shaft speed sensing probes.
- Single valve multistage steam turbines have successfully used pneumatically operated servomotors and trip and throttle valves several times. It is important to note that since magnetic bearings require pneumatic cooling, the use of a pneumatic actuator and trip and throttle valve does not introduce a different utility system in place of the oil system.
- Dry couplings have been used in place of oil lubricated gear type couplings in turbomachinery strings for many years.

These new technologies, verified by several years of operating experience, now allow us to implement a completely oil-free turbomachinery string.

DESCRIPTION OF THE TURBOMACHINERY STRING

The oil-free turbomachinery string comprises a single valve, multistage steam turbine driving a multistage propane refrigeration compressor, an electronic governor and overspeed trip device, and a complete magnetic bearing control system.

Steam Turbine

As shown in Figure 2, a single valve, four stage steam turbine drives the compressor and is rated at 2464 hp at 9088 rpm. The rated steam pressure is 4500 kPa absolute (652 psia) at 420°C (788°F), and the turbine exhaust pressure is 530 kPa absolute (76.7 psia).

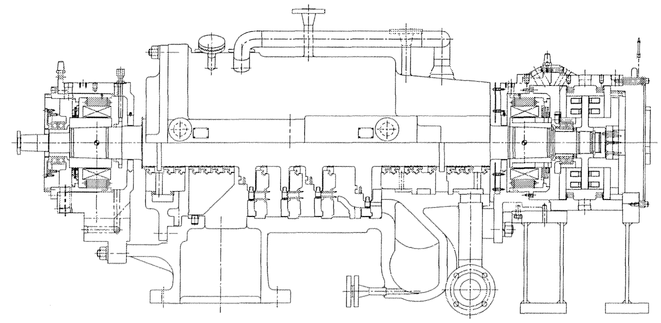

Figure 2. Steam Turbine Cross-Section.

The selection of the steam turbine's internal components, including the rotating and stationary blades, diaphragms, shaft sleeves, and interstage seals, was based on desired performance characteristics and mechanical integrity, and not on the use of magnetic bearings. The design of the magnetic bearings dictates the use of a hydraulically fitted thrust collar, since it must be removed for the assembly and disassembly of the bearings.

The steam turbine's combined trip and throttle valve utilizes a pneumatically operated trip cylinder. The location of the string, outdoors in a semiarid environment subject to frequent sand storms, required that corrosion resistant linkage materials be used. The movement of the steam inlet valve is controlled by a pneumatic actuator with an electronic to pneumatic transducer operating on a 4 to 20 mA signal from the electronic governor. The steam turbine's governor is integrated with the compressor's antisurge system and is installed in the petrochemical complex's control room. Three magnetic speed sensing probes furnished the signals for the two out of three voting logic electronic overspeed trip device. These are all standard components and their selection was not dependent on the use of magnetic bearings.

The steam turbine's packing glands are sealed using a traditional steam leakoff arrangement consisting of an ejector and gland condenser. Consideration was given to installing dry gas face seals in place of this system; however, since neither the owner nor the original equipment manufacturer had extensive experience with this arrangement, the decision was made to utilize the traditional system.

The steam turbine's casing supports had to be modified to accommodate the large size of the magnetic bearings, but the standard centerline support method that allows equal vertical and horizontal thermal growth from the string's center and axial growth away from the drive end was maintained. During testing, a resonance in the steam end bearing support was detected and required further modification to that support to eliminate the resonance.

An important feature of the magnetic bearing installation was the commonality of spare parts between the steam turbine and the compressor. To this end, identical radial bearings were used on both units. The steam turbine did require a slightly larger axial clearance between the thrust bearing and the thrust collar to allow for the unit's greater thermal growth. This was accounted for by a difference in the shim thickness used to set the rotor's axial movement, allowing identical thrust bearings to be installed in both the compressor and steam turbine.

Compressor

As shown in Figure 3, the seven stage centrifugal compressor was designed for refrigeration service. The selection of the impellers, diaphragms, interstage shaft seals, and sleeves was, as in the design of the turbine, based on aerodynamic performance and mechanical considerations rather than the use of magnetic bearings. Again, the magnetic bearing design required a hydraulically fitted thrust collar to facilitate assembly and disassembly.

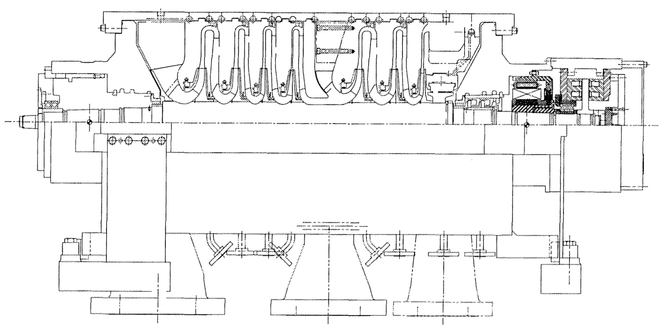

Figure 3. Compressor Cross-Section.

The compressor uses tandem dry gas face seals to contain the process gas. This type of seal requires a dry gas seal buffer skid to supply clean filtered gas at the proper supply pressure. The compressor discharge is used as the source of the seal gas. An additional air buffer was furnished to prevent migration of the process gas into the bearing area from the secondary seal.

The compressor is supported on the foundation by a standard centerline wobble-foot support system. This system allows thermal growth away from the axial centerline, while the wobble-foot feature permits growth away from the drive end to minimize the coupling deflection. No special considerations were made in the casing support design to account for magnetic bearings.

A dry diaphragm type coupling transmits power from the steam turbine to the compressor. Both the owner and the equipment manufacturer have had extensive experience with this type of coupling. The coupling selection was not affected by the use of magnetic bearings. The coupling is secured to the compressor and turbine shafts by hydraulically fitted hubs. The coupling guard design was affected by the use of magnetic bearings. The manufacturer of the coupling guard considers surface temperature in the design and makes provisions to minimize this for personnel safety. Normally, one option to reduce high coupling guard surface temperatures is the introduction of cooling oil from the lubrication system. In an oil-free string, that option has been eliminated. Special consideration must therefore be given to the coupling guard design. In the design of the guard for this particular string, measured surface temperatures of guards of similar design, operating under similar conditions, were compared with predicted surface temperatures ensuring that high surface temperatures would not be encountered.

Magnetic Bearing Controls

Control Loop

Many papers have been written about magnetic bearing operational theory (DeChoudhury, et al., 1993; Destombes and Allaire, 1995; and Wagner, 1988). A review of the different types of magnetic bearings concludes that active magnetic bearing systems are more readily accepted, due to higher load capacity and controllability. The ability to actively control the characteristics of a magnetic bearing system provides rotordynamics experts with significant advantages and offers a new approach toward solving problems. However, along with these advantages, it is important to understand how to apply this technology and its limitations.

Active magnetic bearings are inherently unstable because electromagnets have a negative stiffness. The control loop required to stabilize the system also determines the rotordynamic operation of the machine. Figure 4 shows the closed control loop for one bearing axis that is made up of the position sensors, signal conditioning system, power amplifiers, and bearing electromagnets. The control loop provides the appropriate output response of the electromagnets based on a rotor position input signal. This ratio of output to input is known as the controller's transfer function. It is important to note that the shaft is an important component of the closed loop system. The components of the control loop and their functions are described in the following sections.

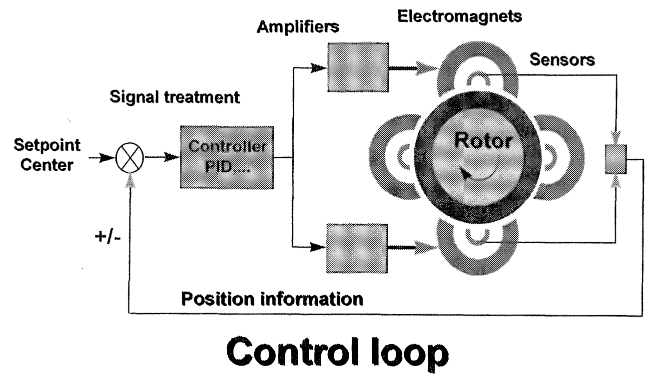

Figure 4. Closed Loop Control System.

Position Sensors

The sensors provide continuous shaft position feedback that is compared with a referenced (centered) position. Any shaft motion away from the referenced position will result in an error signal being sent to the signal conditioning system. The variable inductive type position sensors are insensitive to temperature changes and have a high signal to noise ratio. A pair of opposing radial sensors, which are mounted as close to the bearing stators as possible, provide control for one radial axis or one degree-of-freedom. Each machine has five degrees-of-freedom, four radial and one axial. A special slip sensor is used to sense axial position and it is combined with the radial sensors on the thrust end of each machine. This arrangement provides an accurate measurement of axial position at a location very near the thrust bearing actuator.

Signal Conditioning

The signal conditioning system is the heart of the magnetic bearing control, because it determines the magnetic bearing characteristics. The error signal generated by the position sensors is conditioned by applying gain and phase, which are electrical properties equivalent to stiffness and damping. This transfer function is similar to a mechanical bearing system that provides stiffness and damping based on the mechanical geometry of the bearing and oil properties. The controller transfer function results from a proportional-integral-derivative (PID) control algorithm that determines the proper bearing characteristics required for stable operation. The transfer function for the steam turbine is shown in the Bodé plots (gain and phase versus frequency) of Figure 5. The transfer function can be analyzed to show the contribution of each part of the PID control loop. At low frequency (up to 10 Hz), there is a very high gain or stiffness resulting from the integration of the sensor signal over time. This high stiffness at very low frequencies is necessary to keep the rotor centered if the static forces change. Bearing stiffness that is proportional to the shaft displacement is defined as the minimum system stiffness occurring at 15 Hz. In the operating speed range of 120 to 160 Hz, there is a positive phase due to the differentiation of the position signal over time. This means that damping is present when displacement amplitudes are high due to the shaft critical speeds. Finally, the gain is rolled off and the bandwidth of the controller limited so that higher frequency resonances cannot be excited by the system.

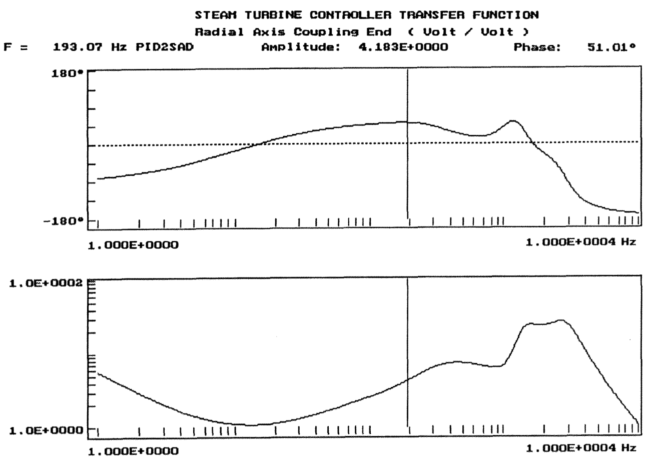

Figure 5. Bodé Plot of Controller Open Loop Transfer Function.

The gain curve shown in Figure 5 represents the bearing dynamic stiffness (K), defined as the vector sum of the real stiffness (k) and the imaginary stiffness known as damping. The phase curve shows the phase angle difference between the controller input and output defining the proportion of real stiffness to imaginary stiffness. Using these relationships, we can identify the bearing characteristics in the same stiffness and damping terms as a conventional bearing (Pinckney and Keesee, 1991).

$$k = K \cos (p)$$
$$B = K \sin (p) / \omega \tag{1}$$

Where:

k = Real stiffness, N/mm (lb/in)
K = Dynamic stiffness, N/mm (lb/in)
p = Phase angle, degrees
B = Damping coefficient, N-sec/mm (lb-sec/in)
ω = Frequency, rad/sec

It is important to note that the magnetic bearing characteristics are functions of a frequency-dependent transfer function. This means that the characteristics are dependent on the frequency of excitation, which is quite different from conventional bearings where the characteristics are based primarily on shaft speed for a given bearing geometry.

Power Amplifiers

Once the position sensor's error signal has been conditioned, the bearing electromagnets require a current source to center the shaft. This current is provided by a power amplifier, one per magnet, operating at 120 VDC that can supply up to 60 A current to the bearing coil. The power amplifiers are pulse width modulated (PWM) type, which were selected due to their efficiency and high bandwidth.

Bearing Actuators

The bearing actuators are electromagnets that generate attractive magnetic forces to center the shaft in a small air gap. The journal magnetic bearing consists of two pairs of opposing electromagnets that control two degrees-of-freedom. The radial bearing is made up of a stack of laminations with coils configured to form alternating magnetic poles. Laminations are also mounted on the shaft to minimize hysteresis and eddy current losses.

Shaft

The location of the shaft, which is the final component of the control loop, is controlled by the bearing actuators based on the controller's transfer functions. Since the shaft closes the loop, it is important to accurately identify the shaft critical speeds and mode shapes so the transfer function can provide the bearing characteristics necessary for stable operation.

Control Philosophy

The control algorithm's design is vital because it determines the range of stable operation. A rotordynamic analysis is performed and a predetermination of the control algorithm is made. Because the actual shaft closes the control loop, it is important to develop an accurate rotor model so that the predefined transfer function provides an acceptable response, and controller tuning is kept to a minimum.

There are two important criteria that must be considered in the control algorithm's design: first, the control algorithm must provide sufficient gain and phase margins on all resonant frequencies to ensure the system's stability. The control algorithm must account for all frequencies within the controller's bandwidth, not only the ones within the operating speed range. Typically, the designer must pay close attention to the stability of at least the first four or five bending modes. Second, the controller design must achieve proper stiffness and damping characteristics for all frequencies up to the maximum operating speed to comply with vibration, amplification factors, separation margin from critical speeds, and unbalance response specifications. The controller must also result in a machine insensitive to transient process conditions such as surge or stonewall, or more steady excitations such as unbalance or rotational harmonics. As shown in Figure 5, there is positive phase at the resonant frequencies crossed during machine acceleration. The rotor mode shapes at these resonant frequencies also determine the damping effectiveness based on the amount of shaft deflection at the bearing and sensor locations.

Description of the Bearing Control System

One of the two identical control cabinets that house the controls for the turbine and compressor is shown in Figure 6. The system converts the incoming 480 VAC power to the 120 VDC required for the operation of the bearings. In addition to the signal conditioning system, the panel contains a microprocessor monitoring system that provides bearing parameter information for user interface. A backup battery, capable of furnishing sufficient DC power for up to 15 minutes to facilitate a controlled shutdown of the string if AC power is lost, is also included.

ROTORDYNAMICS

In order to assure that this unique string, where both the driver and the compressor are supported by magnetic bearings, has satisfactory vibration characteristics, a highly detailed rotordynamic analysis was conducted. Although the rotordynamic analyses of the compressor and the turbine were initially conducted individually, they ascertained that there were no undamped mode characteristics at the critical speeds or near the operating speed, and that the rotors did not have node points at or near the probe location allowing adequate control of vibration by the magnetic bearings (DeChoudhury, 1995). The nature of the vibration characteristics was also evaluated to determine the rotor sensitivity to

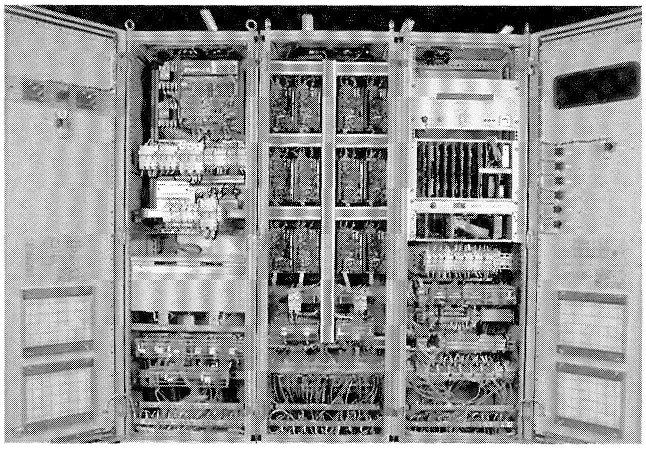

Figure 6. Control Panel.

unbalance and to ensure that any peak response at or near the operating speeds was well damped. This was achieved through evaluation of amplification factors.

Since the effect of the magnetic bearings on a coupled string was unknown, it was decided to conduct a train rotordynamic analysis to ascertain satisfactory vibration characteristics. The results of this analysis are presented here along with a comparison of the prediction with test data. In order to perform the train rotordynamic analysis, each rotor was modelled using the "lumped masses connected by a weightless spring" model (Lund and Orcutt, 1967). The coupling hubs were assumed to be integral with the shaft ends and the remainder of the coupling was modelled as an interconnecting hollow shaft. The diaphragm coupling was modelled as a shaft connecting the rotors with the flexible diaphragm replaced by cylindrical sections of equal length, having the same bending stiffness as the diaphragm.

Figure 7 shows the rotor bearing arrangement where the radial bearing span of the compressor is 1689.4 mm (66.51 in), and the steam turbine is 1630.2 mm (64.18 in). The distance between the shaft ends of the compressor and the steam turbine is 762 mm (30 in). The major shaft diameter of the compressor under the impellers is 1485.8 mm (8.5 in) and the compressor shaft is scalloped on both sides of each impeller to improve the lateral shaft rigidity, as much as possible, while meeting aerodynamic requirements. The steam turbine main shaft diameter is 181 mm (7.125 in) and the turbine disks are integral to the shaft, which increased the shaft lateral stiffness. The auxiliary bearings, or run down bearings, are located outboard of the journal bearings and inboard of the thrust bearings. This was done to reduce the rotor's flexibility during coastdown on the auxiliary bearings.

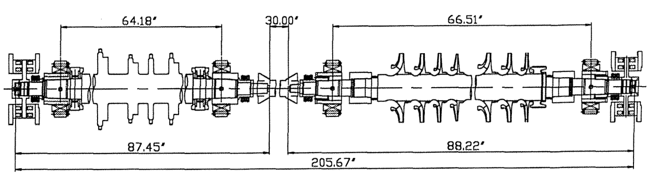

Figure 7. Turbine-Compressor Rotor Bearing System.

The frequency-dependent rotordynamic characteristics of each radial bearing are presented in Table 1. Using the magnetic bearing stiffness characteristics, the undamped critical speeds and mode shapes associated with those frequencies are shown in Figures 8 and 9. The rotor string was modelled from the steam turbine to the compressor, from left to right. It can be concluded from the undamped natural frequencies and their mode shapes that the node points, except for the higher modes, are well away from the bearing centerlines. These higher modes are sufficiently removed from the string's maximum speed to be of no concern. The damping would thus be effective in controlling the rotor vibrations within the operating speed range, and well damped peak responses were to be expected.

Table 1. Rotordynamic Coefficients.

Frequency (HZ)	Compressor				Turbine			
	Thrust Side		Coupling Side		Thrust Side		Coupling Side	
	Stiffness (lb./in.)	Damping (lb.s/in.)	Stiffness (lb./in.)	Damping (lb.s/in.)	Stiffness (lb./in.)	Damping (lb.s/in.)	Stiffness (lb./in.)	Damping (lb.s/in.)
10	346000	-2700	142000	-830	138000	-852	154000	-791
50	371000	444	192000	488	182000	484	201000	545
75	403000	504	224000	464	216000	473	237000	525
100	443000	508	250000	449	254000	453	270000	500
125	496000	501	279000	453	295000	433	304000	484
150	550000	482	324000	468	340000	413	342000	471
200	672000	437	522000	480	438000	373	447000	455
250	796000	389	820000	381	547000	337	596000	426
300	915000	345	1010000	230	651000	300	774000	375

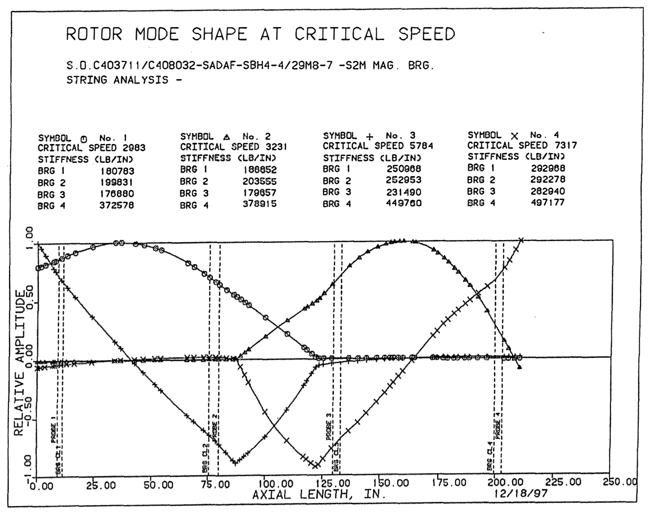

Figure 8. Rotor Undamped Critical Speeds and Mode Shapes.

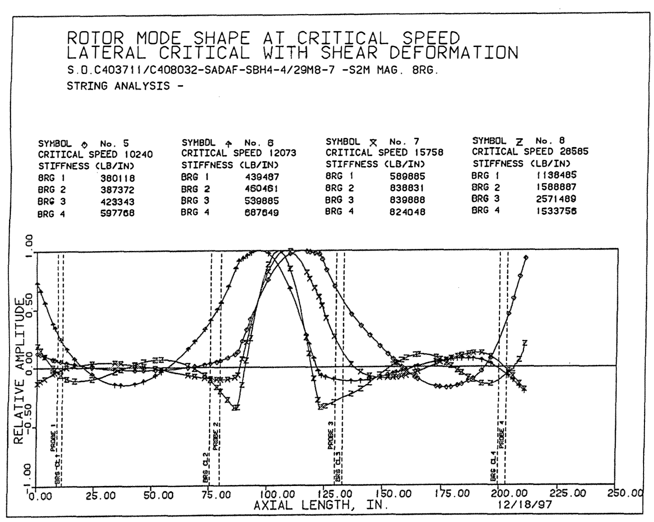

Figure 9. Rotor Undamped Critical Speeds and Mode Shapes.

Table 2 presents a summary of the rotordynamic results obtained from the string analyses. Results are presented for the rotor bearing systems supported on magnetic bearings and also supported on the auxiliary bearings. The rigid support critical speeds presented in Table 2, along with Figures 8 and 9 showing the undamped modes, provide overall dynamic characteristics of the rotor bearing system relative to the operating speed. When the rotors are supported on the magnetic bearings, it can be noted that there are two peak responses below the operating speed; both peak responses are well damped and do not affect the system. The results of the log decrement analysis, as shown in Table 1, corroborate that stable operation without subsynchronous vibration problems may be expected.

Table 2. Summary of Rotordynamic Analysis.

	RESULTS ON MAGNETIC BEARINGS				RESULTS ON AUXILIARY BEARINGS			
ITEM	TURBINE		COMPRESSOR		TURBINE		COMPRESSOR	
RIGID SUPPORT CRITICAL SPEEDS								
1	6333		4730		4417		3196	
2	21655		14572		17177		10701	
3	26456		21999		>30000		23678	
UNBALANCE PEAK RESPONSES (AF)*								
	Thrust End	Cplg. End	Cplg. End	Thrust End	Thrust End	Cplg. End	Cplg. End	Thrust End
1	3100(1.67)	3000(1.96)	3100(1.72)	-	4250(12.66)		3000(6.87)	
2	7650(1.45)	-	7300(1.30)	-	14700(23.02)		10100(29.65)	
LOG DECREMENT STABILITY ANALYSIS								
	1.102		0.519		-		-	

*** Peak Response in rpm, amplification factor (AF) in parentheses.**

Attention must also be paid to the evaluation of the dynamic behavior of the rotor during coastdown on the auxiliary bearings. The auxiliary, or run down bearings, are ball bearings supported by Borelli rings that are expected to provide a flexible mounting with structural damping. The coastdown of the units on the auxiliary bearings is a transient rotordynamics problem. In the absence of proven analytical tools to simulate the transient response of such a complicated flexible rotor bearing system, steady-state rotordynamic analyses were performed to evaluate the rotor characteristics. The stiffness characteristics of the ball bearing were used along with their estimated structural damping (Jones and McGrew, 1965). The results of these analyses for the steam turbine and the compressor are shown in Table 2, and indicate that both rotors pass through a critical speed with a relatively high amplification factor. The second critical speed for both the steam turbine and the compressor have very high amplification factors. However, they are located at much higher speeds than the maximum continuous speed and will have no effect on the rundown characteristics.

TESTING

Control System Tuning

Tuning was performed at the OEM facility after assembly of the turbomachinery string. The first step was to levitate the shaft with a controller whose characteristics have been predetermined as a result of the rotordynamic analyses, and then verify the calibration (sensitivity) of the position sensors, verify the bearing and auxiliary bearing air gaps, and identify the system resonant frequencies. One advantage of magnetic bearings is that they can be used to shake the shaft for the identification of the various transfer functions. The second step was to verify the accuracy of the model by comparing the measured transfer functions with predicted ones. The actual transfer functions were then fed back into simulation software with the string model, allowing the designer to predict and quantify the system stability. Using this simulation model, the PID controller parameters may then be modified and optimized by iteration until satisfactory damping and stiffness characteristics are achieved. The third step was to implement the optimized PID parameters in the actual controller and measure the transfer functions to assess the gain and phase margins. Since the controller tuning was done with the string at a standstill, it was impossible to measure the gyroscopic effects at the various critical speeds. However, these effects were verified during the rotational tests that followed and resulted in the optimal tuning of the controller based on measured unbalance responses and resonant frequencies.

Mechanical Running Test

Except for the initial steam turbine test that verified the proper operation of overspeed trip mechanism, all other tests were conducted on the complete string. The test stand's electronic governor was used to control speed during all running tests, since the string's governor is an integral part of the overall control system and was not available for any of the factory acceptance tests.

The string mechanical test was conducted by operating the compressor at approximately 50 percent load, which was attained by pressurizing the test loop with carbon dioxide. The mechanical test lasted four hours at the string's maximum continuous speed of 9543 rpm, followed by 15 minutes of operation at the trip speed of 10,497 rpm. During the stand-alone turbine test, that unit was operated for 10 minutes at 110 percent of trip speed (11,546 rpm). An unbalance response test conducted on each individual rotor consisted of acceleration of the unit from rest to maximum continuous speed with an unbalance weight attached to the coupling.

The overall vibration plots versus speed for the steam turbine and the compressor are shown in Figures 10 and 11. In the plot for the steam turbine, Figure 10, well damped peak responses were observed at 3100 rpm at the coupling end and at 3400 rpm at the thrust bearing end, with no second peak responses observed. The vibration plots for the compressor, Figure 11, indicate that the first well damped responses at both the thrust bearing end and coupling end occurred at 3000 rpm. No second peak response was observed at the thrust end; however, a broad peak at 9000 rpm was observed at the coupling end. The observed well damped first peak responses correlate well with the prediction, but the second peak response's location was not as predicted. The well damped nature of the vibration characteristics was expected and maintained throughout the operational range.

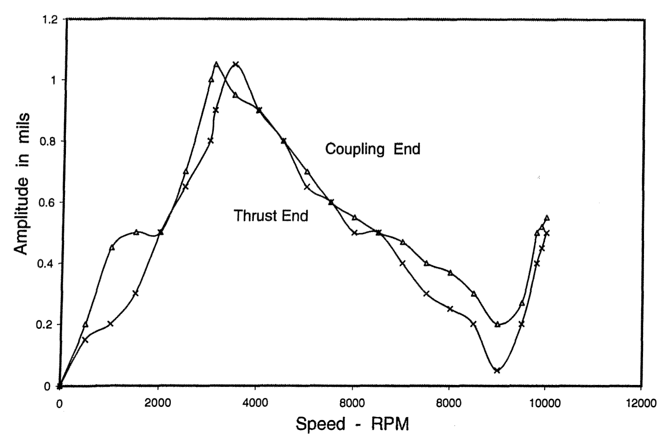

Figure 10. Speed Versus Observed Overall Amplitude at the Turbine in Vertical Direction.

Two "drop" tests—where the rotors are suddenly delevitated and allowed to coast down—were conducted on the auxiliary bearings. Figure 12 shows the amplitude of vibration as a function of time when the compressor rotor was allowed to coast down on the auxiliary bearings. Since the entire compressor drop test lasted approximately 7.5 seconds, Figure 12 is an expanded time base

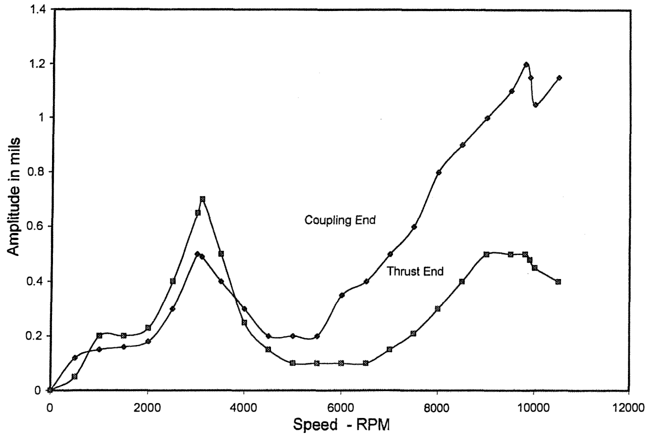

Figure 11. Speed Versus Observed Overall Amplitude at the Compressor in Vertical Direction.

plot for the drop portion of the compressor test. There was a buildup of 4600 cpm vibration in about 0.07 seconds. This subsynchronous vibration quickly decayed and was mainly synchronous at around 1.5 mils. The steam turbine drop test lasted about 8.7 seconds, with the coupling end, Figure 13, showing about five cycles of 3100 cpm vibration that quickly decayed from 3 mils to a very low value. The overall vibration level, as shown both in Figures 12 and 13, decayed quite rapidly resulting in a log decrement value for the compressor of 0.08 and for the turbine 0.22. It is interesting to note that when the rotors were delevitated onto the auxiliary bearings during the drop test, the compressor and the turbine essentially responded to their first rigid support speeds for auxiliary bearings as listed in Table 2.

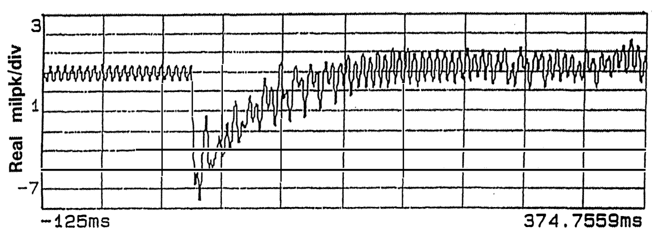

Figure 12. Observed Amplitude of Vibration as a Function of Time in Vertical Direction—Compressor Drop Test—Coupling End.

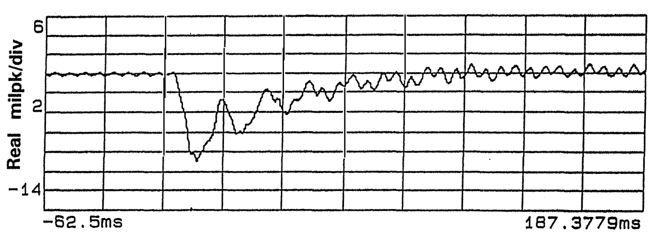

Figure 13. Observed Amplitude of Vibration as a Function of Time in Vertical Direction—Turbine Drop Test—Coupling End.

Aerodynamic Performance Test

In addition to the mechanical tests, the compressor underwent an equivalent Class III aerodynamic performance test as defined by ASME PTC 10-1965. The test gas was Freon 22 and the speed was 7890 rpm. The results show that the compressor is slightly more efficient than predicted at the design point. Of greater importance, from the magnetic bearing point of view, was the fact that the turbomachinery string was operated at varying loads. The test was conducted at approximately 85 percent of design speed, and the test load was about 60 percent of design point power. During the course of the test, the load was varied while holding the speed constant from stonewall to surge, and in order to establish the surge line, the compressor was surged at 7000 rpm and 8500 rpm. The magnetic bearing control system was able to maintain stability during all these load changes, including surge conditions at three separate speeds.

OPERATING EXPERIENCE

After installation at the site in Saudi Arabia, the turbomachinery string was operated for two weeks in February 1997, three weeks during July, and finally went online in November 1997. As of this writing, the string has only had about three months of online operation. Process demand has not required the compressor to operate at speeds higher than the minimum governor speed of 7270 rpm. The string has been operated for short periods of time at higher speeds by forcing the recycle kickback valves open and manually increasing speed. Obviously, unattended operation for long periods at this condition is not prudent. The compressor is currently running very smoothly, with vibration levels at approximately 17.8 μm (0.7 mils) and appears able to meet the unit process demand at the minimum governor speed. The turbine is running slightly rough, with measured vibration levels of approximately 50.8 μm (2.0 mils). This unit is believed to require field balancing.

During steady-state operation, the turbine speed varies by plus or minus 200 rpm, and the turbine appears to be sensitive to load changes, with slight changes in load resulting in speed swings. Attempts were made to correct this instability by modifying the electronic governor control loop. The problem has been temporarily corrected by reducing the steam inlet pressure.

The plant is currently running at less than expected capacity and attempts are being made by plant operating and process personnel to bring it up to full capacity. The turbine vibration and governor instability problems do not appear to be related to the magnetic bearings. Current plans call for the string to be shutdown sometime in the spring of 1998. The source of both problems is expected to be thoroughly investigated and permanent corrective action taken at that time.

LESSONS LEARNED

The factory tests of this string took an inordinately long time. The string was on the manufacturer's test stand for approximately five months, which was two months longer than planned. This extremely long residency time was caused by a unique chain of unforeseen events. About halfway through the manufacturing phase of the project, the original magnetic bearing manufacturer decided to discontinue this technology. Several meetings were held between the equipment user, the engineering contractor, the turbomachinery manufacturer, the original bearing manufacturer, and the new bearing manufacturer. A plan was agreed upon and implemented where the original manufacturer would complete the magnetic bearings, since they were near completion at that time, while the new bearing manufacturer would design and build the control system. Although this situation was not ideal, the bearings and control system were delivered on time. The new manufacturer had a slightly different control philosophy than the original manufacturer; however, this was not as important as the fact that these differences were not conveyed to the rotating equipment manufacturer. This led to delays caused by different interpretation of test results, especially during the tuning process. Once it became apparent that results were being interpreted based on different assumptions, all parties agreed to follow the same philosophy and testing proceeded smoothly.

Steam turbines must have the capability to be run separately from their driven equipment so that they may be properly commissioned, both initially and after major overhauls. While this may be obvious to steam turbine manufacturers, engineering contractors, and owners of rotating equipment, it was not previously understood by magnetic bearing manufacturers. This fact became apparent on the test floor when modifications to the control system had to be made to allow turbine solo runs, increasing costs and adding further delays. The system had to be furnished with two sets of electronic control boards, one to be used for string operation and the other for solo operation of the steam turbine. Had this been known at the start of the project, provisions, such as a switch for changing the operating mode, could have been incorporated in the system design.

An open dialogue between the bearing designers, the engineering contractor, the turbomachinery manufacturer, and the ultimate user is vitally important to the success of such a project. Care should be taken to clearly define every aspect of operation in the job specification. Initial coordination meetings must be held in order to ensure that all parties understand and agree upon the design philosophy. At least one design review should be held in order to verify that the design philosophy is being properly implemented, and test procedures should be reviewed in detail and agreed upon by all parties long before any actual testing takes place. Although this may seem obvious, and miscommunication on a project is not unique to magnetic bearings since there are very few detailed specifications defining magnetic bearings, special attention must be paid to communications in any project where they are involved.

CONCLUSIONS AND RECOMMENDATIONS

Good communication is essential for the successful completion of any project, and is especially important in a project involving new technology such as the implementation of magnetic bearings. The magnetic journal and thrust bearings are an integral component of the machine, and open communication between the designers of the bearings and the designers of the rotating equipment is an absolute necessity. During the design of this particular string, the magnetic bearing manufacturer and the turbomachinery manufacturer exchanged detailed machine drawings electronically, which allowed the integration of the bearing drawings into the compressor and turbine drawings without error. This technique proved its value during the assembly of the machines where there were very few assembly problems. This deep involvement in the communication process should be extended to the users as well, so they may gain a complete understanding of the system, while ensuring that their requirements are properly interpreted and implemented.

The good correlation between predicted and measured rotor response has shown that detailed rotordynamics analyses, conducted as early as possible in the design phase, will ensure that the system will operate as expected. It is much easier to make modifications to a drawing than to a piece of hardware.

At present, there is no industry wide standard defining the application of magnetic bearings in turbomachinery. As the use of this technology becomes more popular, the need for industry wide standards will become apparent. Terminology must be defined and standardized. Minimum equipment and documentation requirements should be determined, so that an end user contemplating the use of magnetic bearings will have a good understanding of the basic scope of supply and the industry standards, along with the quality level to which the equipment must be built. While the end user will always have the option to add or to delete from these standards as he sees fit, they will provide him with a place to start.

The purpose of this paper was to discuss the processes followed and the rationale used in the development, design, and construction of a completely oil-free turbomachinery string and to present to the reader the possible pitfalls encountered in the production of this string, with the expectation that future users contemplating such an endeavor will learn from these experiences. Although the problems encountered were discussed in detail, it must be emphasized that the overall project was a success. With the complete cooperation of all parties involved, problems were identified and totally resolved. Although the turbomachinery string has had limited operating time as of this writing, there is no reason to believe that it will not operate exactly as anticipated. It has been demonstrated that the technology, backed by many years of operating experience, is currently available for the implementation of oil-free turbomachinery strings into process service. All of the components, including the magnetic bearings used in the string described in this paper, have been proven by years of operating experience. The oil-free package is simply the culmination of these accepted components. End users, contemplating the design of a new string of turbomachinery, may now consider the use of magnetic bearings as a viable alternative to traditional oil lubricated bearings.

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